A SHELL ECO-MARATHON CONCEPT CAR ENGINE DESIGN

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ABSTRACT

High-power, low weight and ease of fabrication are the key factors the young engineers consider when it comes to their participation in the annual Shell Eco-marathon competition. The competition encourages young engineers to come up with innovative vehicles that make extremely high mileage on a gallon of fuel. The competition allows for a mixed mode driving. The drivers can switch off the engines once a good acceleration has been reached and is enough to coast the vehicle. This can be repeatedly done until the race-circuit is completed. Many of the teams adapt existing engines and build aerodynamic bodies over the engine but others also want to design the engine from the scratch. In this paper, we present a simple design for 40cc engine and introduce a novel concept for an engine without an oil pump specifically suitable for this application. An overhang single cylinder IC engine with a crankshaft length 150mm with 32mm stroke and 40mm bore has been design for the Shell Eco-marathon race.

Keywords: Shell Eco-marathon, concept car, overhanging crankshaft, car engine

INTRODUCTION

Shell Eco-marathon is an annual competition organised by Shell for young engineers to design cars that can consume extremely low amount of fuel to cover great distances. Regular cars make just about 50 miles per gallon but the vehicles in this challenge reach around 2500 miles per gallon. There are two categories of this competition; the Prototype category or the Urban Concept category. The prospective participating teams are allowed to enter into either of these. In the prototype category, the allowed maximum vehicle weight without the driver is 140kg and a frontal cross-section of 130x100cm and maximum length 350cm. The teams are allowed to design them to be aerodynamic but within the specifications. The urban concept category regulates that the frontal height be 100–130cm and a width of 120–130cm with a total length of 220–350cm and maximum weight excluding the driver to be 205kg. The Shell-Eco marathon concept cars look like the regular passenger cars. Figure 1 shows a participating team in the concept car challenge.

An important factor in the design of these vehicles will include how to lower the overall weight and to increase engine power. The road resistance depends, linearly, on total mass of the vehicle and the driver and the square of the velocity in the drag term. Adeniyi& Mohammed (2012) indicated that the high mileage attributed to these vehicles is partly contributed from the driving pattern. Most of the teams purchase small Honda engines of the GX series and build the vehicles around it. Some teams desire to build the engine from scratch to get more involved.

This paper presents a simple design for the body of a light engine with simple overhang crankshaft and no oil pump. This can be fabricated in a small workshop and a standard 40mm bore engine head can be fitted or designed.

Figure 1.Concept Car –Winner (ITS, 2012)

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ENGINE DESIGN ANALYSIS

The presented design is for a 40cc engine capacity, 1.50kW and a target 3000miles per gallon and brake specific fuel consumption (BSFC) of 0.199 kg/kWhr and a fuel consumption rate of 0.58 litre/hr based on the work of Adeniyi(2008). The specifications of the parts are shown in Table 1.

Table 1.Engine Specifications

Part-description	Dimension (mm)
Big-End diameter	10
Small-end diameter	10
Con-rod length	72
Bore	40
Stroke	32
Crankpin diameter	10
Crankshaft length	150
Crankshaft Diameter	20
Bearings	20 int. dia., 32 outer dia. 8 thick, 4 No. Bearings
Pulley	22mm dia.

Connecting Rod

The maximum pressure in the cylinder is 30MPa. The maximum force exerted on the connecting, F_{conn}(N), rod is experienced at the top dead centre is given in equation (1)

$$F_{conn} = P_{max} A_{pc} \qquad (1)$$

Where A_{pc}= the piston crown area (m^2).

Crankshafts can be either of split-crank or overhang style. To allow for easy servicing or fitting, the overhang crank is recommended. Figure 2 and Figure 3 show the styles of crankshaft design for a single piston engine.

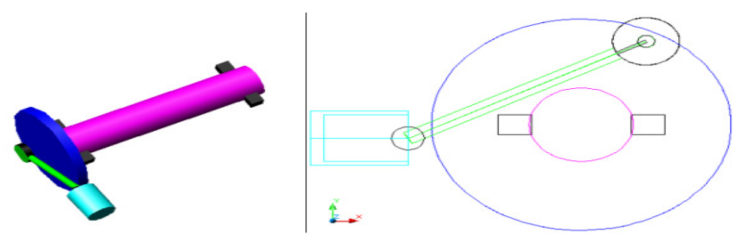

Figure 2.Overhang Crank -Piston on side

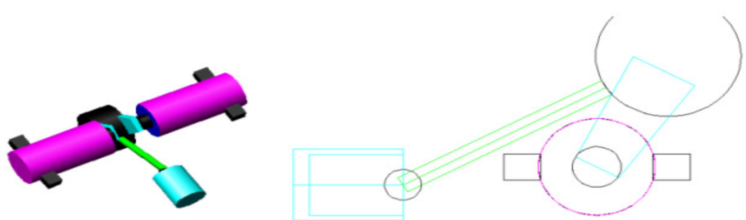

Figure 3.Split Crank -Piston within

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The operating speed is 5000rev.per minute maximum. Silver steel, $E=2.07\times10^{11}$Pa, is recommended for the connecting rod. A check for the maximum force, F_{maxcon} by failure using the Euler buckling is given in equation (2) as the connecting rod is the overhang style.

$$F_{maxconn} = \frac{\pi^2 EI}{(Kl)^2} \qquad (2)$$

Where $K=1/\sqrt{2}$ column factor for column fixed at one end.

Big End Analysis

Figure 4 shows the crankshaft and the connecting rod. The big end connects the con-rod and crankshaft via a crank pin.

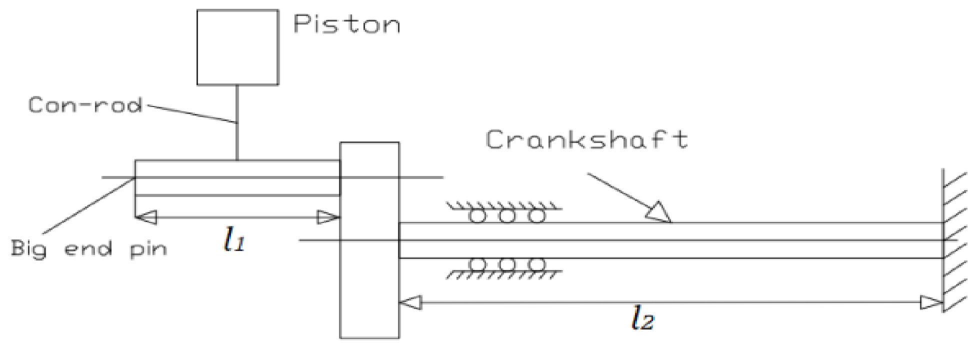

Figure 4.Crankshaft model

The big end pin is modelled as shown in Figure 5. The deflection, w(x), of the big end pin is given by equation (3) and the maximum, w_{max}, occurs at $x=l_1$ in (4). Where p is the con-rod force per unit length and D_p is the big end diameter.

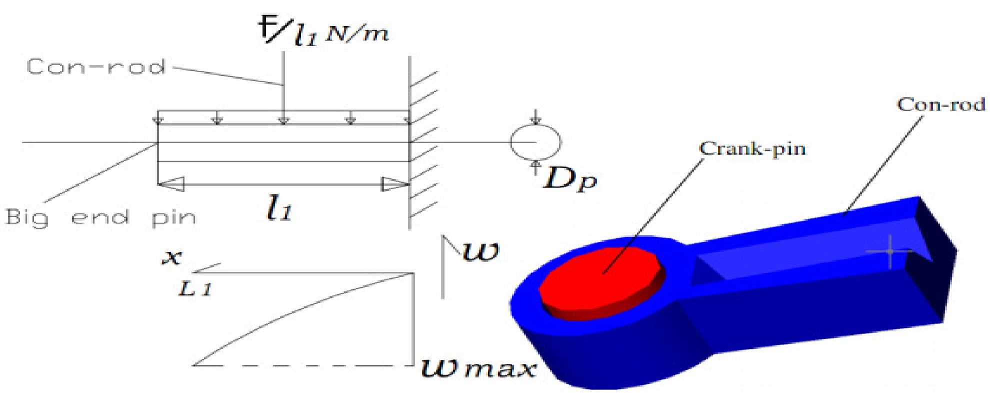

Figure 5.Big pin model

$$w(x) = -\frac{px^2(6l_1^2 - 4xl_1 + x^2)}{24EI} \qquad (3)$$

$$w_{max} = -\frac{pl_1^4}{8EI} \qquad (4)$$

The maximum shear stress can be shown to be given by equation (5).

$$\sigma_{max} = \left(\frac{16}{\pi D_p^3}\right)\left|\frac{pl_1^2}{2}\right| \qquad (5)$$

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Crankshaft Loading

Figure 6 represents an exaggerated deflection of the crank shaft at maximum loading conditions. The deflection at the big end is y_1 and the maximum deflection between the bearings is y_2, in (m). R_1 and R_2 are the reactions, in (N), at the bearings. L is the shaft length (m) and a is the overhang distance from bearing 1.

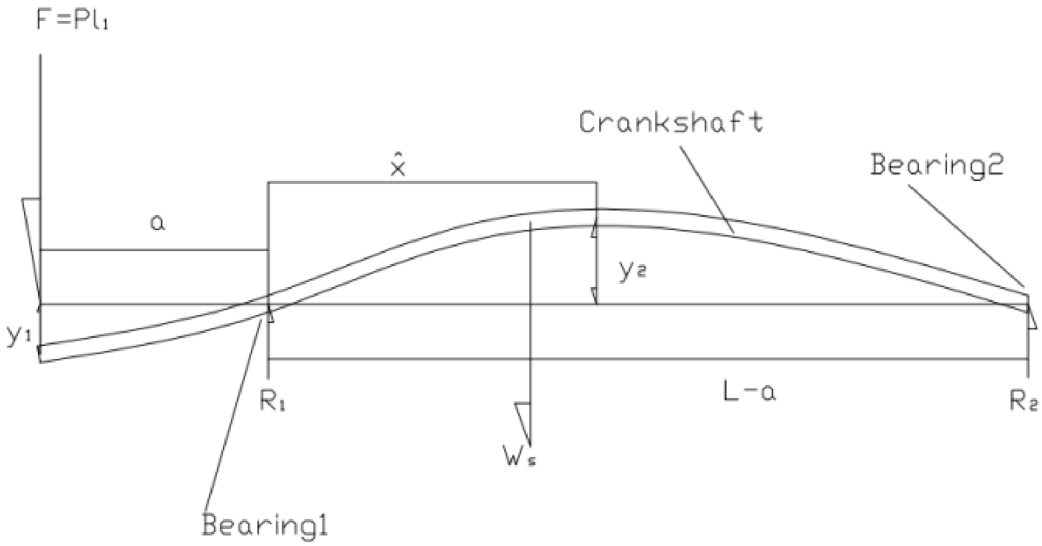

Figure 6.Exaggerated crank deflection

$$y_1 = \left(\frac{\pi D_p^3}{32}\right)\left|\frac{Pl_1 a^3}{3E}\right| \qquad (6)$$

$$y_2 = \left|R_2\left(\frac{\pi D_p^3}{32E}\right)\left(\frac{\hat{x}^3}{6} - \frac{L}{2}\hat{x}^2 + A\hat{x} + B\right)\right| \qquad (7)$$

Where $\hat{x} = L + \sqrt{\left(L^2 - La^2 + \frac{1}{3}a^3 + \frac{2}{3}L^3\right)}$

$A = \frac{3La^2 - 2L^3 - a^3}{6}$ and $B = \left(\frac{L^3}{3} + \frac{1}{3}L^4 - \frac{1}{2}L^2a^2 + \frac{a^3L}{6}\right)$

$$R_1 = mg + pl_1 - \frac{mg\left(\frac{L}{2} - a\right) - \frac{pl_1^2}{2} - pl_1 a}{L - a} \qquad (8)$$

$$R_2 = \frac{mg\left(\frac{L}{2} - a\right) - \frac{pl_1^2}{2} - pl_1 a}{L - a} \qquad (9)$$

Where m= mass of the shaft (kg).

Shaft Torsion

The shaft torsion can be estimated similar to Jones (1989) as shown in equation (10) and for a 20mm diameter shaft, this gives 873.36Nm.

$$T = \left(\frac{\pi D_s^3}{32}\right)\tau_{max} \qquad (10)$$

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Where τ_{max} is the maximum shear stress (N/m).

Bore – Stroke Analysis

The bore stroke is selected using gas leakage, friction and heat loss (Adeniyi, 2008). These relationships are used in simulating the bore-stroke dependence of Figure 8.

Engine Bore stroke relationship

If bore to stroke ratio is defined as r=B/S, and given an engine with a 40cc swept volume, the bore size, B (m) is related to the bore-stroke ratio as given in equation (11). For a square engine, r=1. A short-stroke (over-square) engine has r>1 while a long-stroke (under-square) engine has r<1.

$$B = \sqrt[3]{\frac{160r}{\pi} \times 10^{-6}} \quad (11)$$

Gas leakage

Gas leakage can only occur from the piston rings. The leakage is directly proportional to the piston perimeter, B, as given in equation (12).

$$\text{Leakage} \propto \pi B \quad (12)$$

Friction

Friction is considered as a proportion of the surfaces rubbing, or otherwise, the surface to volume ratio as given in equation (13), where S is the stroke length.

$$\text{Friction} \propto \frac{\text{Surface}}{\text{Volume}} \propto \frac{\pi BS}{40cc} \quad (13)$$

Heat loss

Heat loss is a function of the "exposed" area as given in (14).

$$\text{Heat loss} \propto \frac{\pi BS + \frac{\pi B^2}{4}}{40cc} \quad (14)$$

Volumetric Efficiency

Heywood (1988) defines volumetric efficiency,η_0, of an IC as the ratio of the air mass flowing into the cylinders of the engine from the intake manifold to the theoretical mass of air present in the cylinders at the manifold temperature. The filling-emptying model, Nicolao et al (1996), uses the conservation of mass at the intake manifold.

$$\dot{m}_a = \dot{m}_{at} - \dot{m}_{ac} \quad (15)$$

Where $\dot{m}_a$ is air mass flow rate (kg/s) between the throttle valve and the inlet ports and using the ideal gas equation, it can be expressed as in equation (16) and $\dot{m}_{at}$ is mass flow rate of air (kg/s) through the throttle plate. It is assumed that the manifold pressure is uniform and the temperature is uniform and constant at the intake manifold.

$$\dot{m}_a = \frac{\dot{P}_m V}{RT_m} \quad (16)$$

The mass flow rate of air (kg/s) into the cylinder, $\dot{m}_{ac}$ is given in equation (17) which is adapted from Heywood (1988), where is ρ_a air density (kg/m^3) and is the η_0 volumetric efficiency and S is the stroke or engine displacement (m).

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$$\dot{m}_{ac} = \frac{N}{120} S\rho_a \eta_0 \qquad (17)$$

Pressure Loss

The losses are modelled using the engine power and pressure losses (Ladommatos, 2007). Friction Mean Effective Pressure (FMEP), P_f (bar), given in equation (18) is defined as the difference between Indicated Mean Effective Pressure (IMEP), P_i (bar) and Brake Mean Effective Pressure BMEP, P_b (bar).

$$P_f = P_i - P_b \qquad (18)$$

The crank power, P_c (kW) is described by equation (19). Where $V_s(m^3)$ is the engine swept volume and N is the engine revolutions per second and η_e is the engine efficiency.

$$P_c = P_b \times V_s \times N \times \frac{1}{2} \times 100 \times \eta_e \qquad (19)$$

The indicated power, P_{ip} (kW), is given by equation (20), where z is 0.5 for a four stroke engine or 1 for a two stroke engine.

$$P_{ip} = P_i \times V_s \times N \times \frac{z}{100} \qquad (20)$$

IMEP is a conceptual constant pressure that will produce the same indicated power on the piston crown over the same swept volume. Brake power, P_{bp} (kW), is the actual power output less frictional losses; therefore the BMEP is also a constant pressure if acting on the piston over the stroke expansion and will produce the same work as measured from the crankshaft as given in equation (21).

$$P_{bp} = 100 \times V_s \times P_b \times N \times z \qquad (21)$$

Novel Concept

In an attempt to make the design as simple as possible, two novel concepts are proposed for the oils system and the transmission of power from the crank to the drive.

Oil System

The engine lubrication can be achieved by the introduction of a spoon-like slinger which dips into the oil sump and scoops oil in such a way that when the piston is at the top dead centre, the oil is delivered on the sleeves at the base of the piston. At a high number of revolutions per minute, a mist of oil is created. This concept means there is no need for an oil pump but requires further investigation.

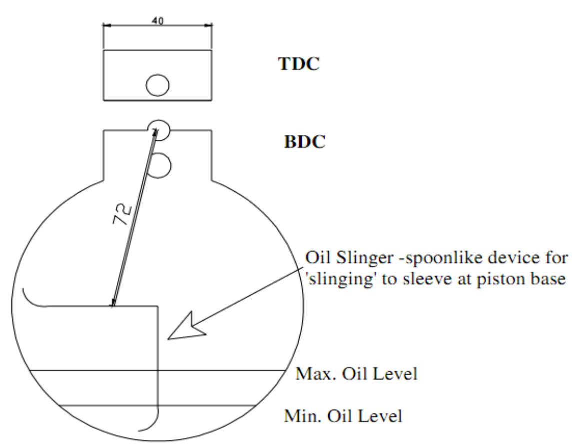

Figure 7.Oil Slinger design

RESULTS AND DISCUSSIONS

Bore Stroke

The power requirements to achieve the driving pattern discussed in (Adeniyi & Mohammed, 2012) could be met using a 40 cc bore. As a comparison, the small GX Series Honda engines have volumetric capacities in this range.

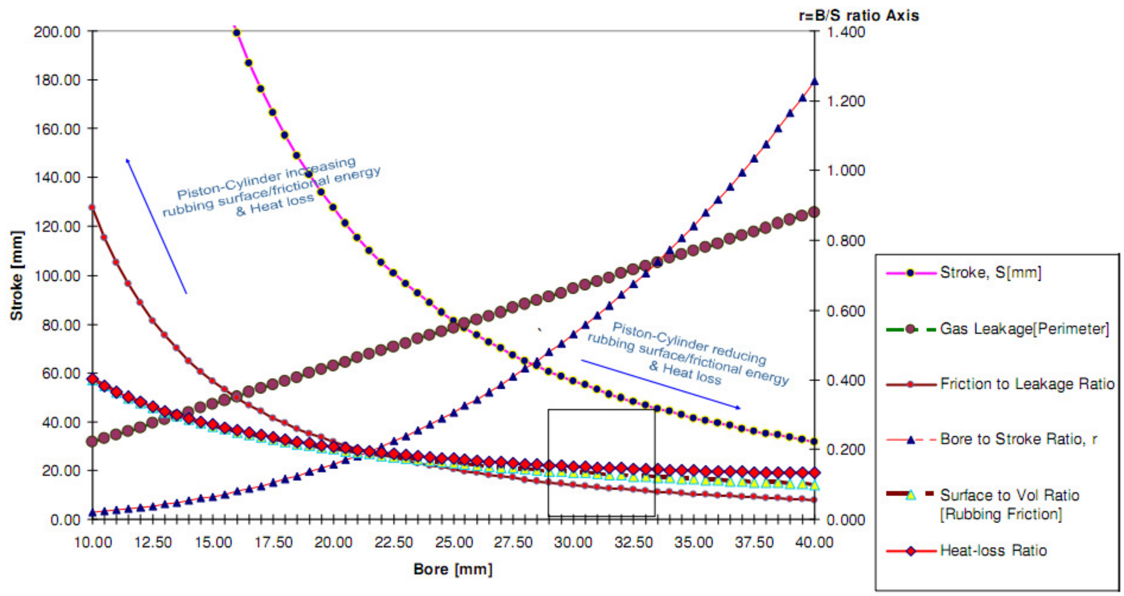

Figure 8.Bore-Stroke dependence of the 40cc engine

Figure 8 shows the bore–stroke dependence for a 40cc engine. From bore diameters of 30mm – 35mm, there is no gain by increasing the bore as the heat loss and the rubbing or reciprocating friction loss are now fairly constant and the gas leakage keeps going up but the effect of gas leakage which in quantitative terms are usually less than 3% of the trapped gas volume per cycle. This gas loss represents only from 2 – 3% loss of power but friction can amount to as high as 10 – 20%. If the piston rings are very tight, the gas leakage effect may not really be as high as predicted so larger bores would be of no help. Settling for a 30mm bore represents a bore to stroke ratio, r, of 30/56 or 0.54 for the 40cc engine. A volumetric efficiency test however shows that the volumetric efficiency starts falling after 3000 RPM for the 30mm bore as shown in Figure 9, and the heat loss is higher as well as indicated in Figure 10. Therefore the 40mm bore is satisfactory.

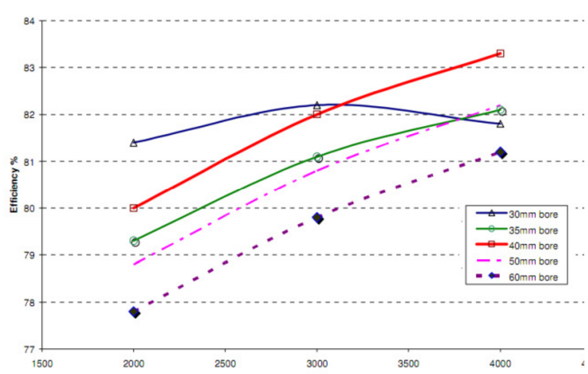

Figure 9.Volumetric Efficiency for different piston bore diameters

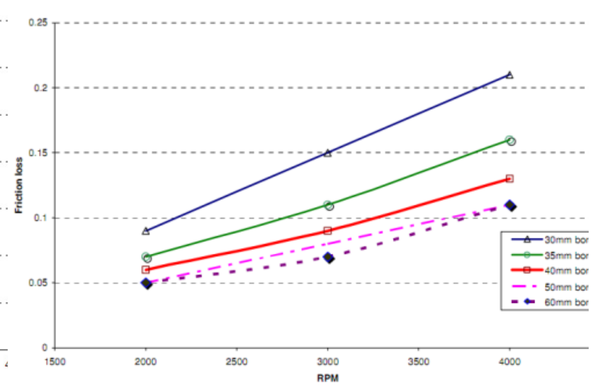

Figure 10.Frictional Losses at several speeds

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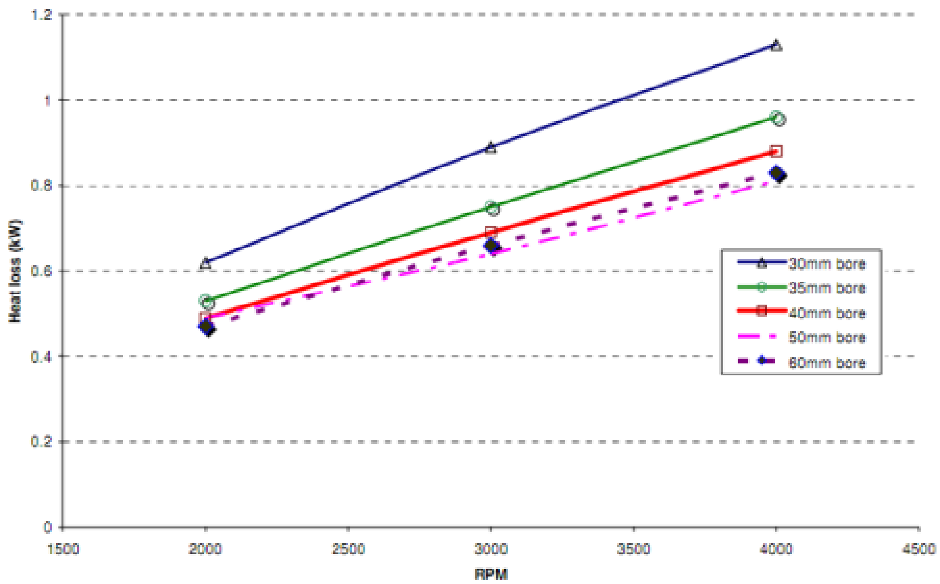

Figure 11.Heat loss for different bores and speeds

Connecting Rod

A long stroke engine has the advantage of high torque or better acceleration which is good for the application but it has the disadvantage of large spatial requirements in the competition. A con-rod to stroke ratio of 1.56 is suggested (Khatiblou, 1996) for conrod with improved fatigue life, however, this will be tiny in this application where more power is desired from little input. For consideration for high speed, vibration and geometric stability, the ratio 2.25 is chosen. From the bore-stroke expression of equation (11), a connecting rod, or conrod, length 72mm will serve the purpose as shown in Figure 12. The figure also shows the connecting rod will not touch the piston sleeve at a 90^{o} crank-angle.

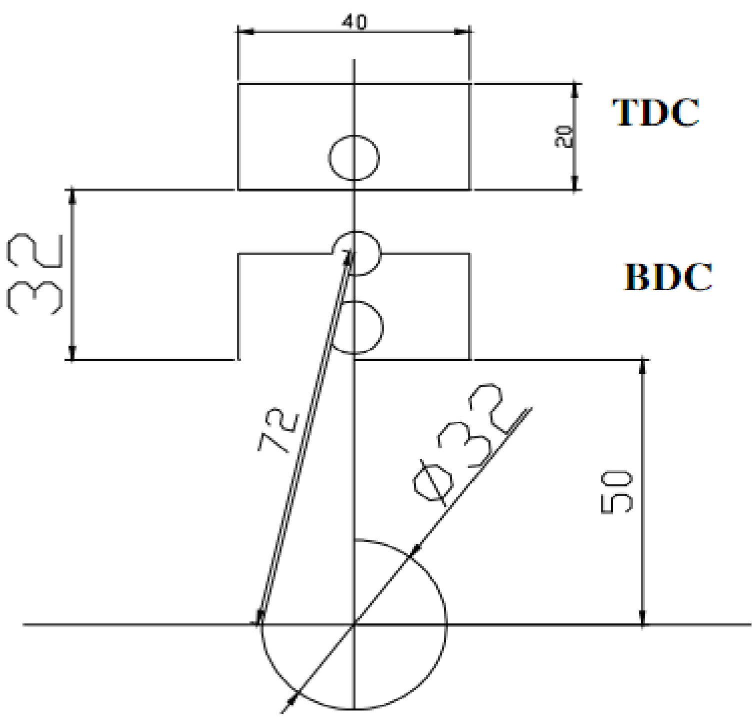

Figure 12.A 2D representation of the Con-Rod and Piston (all in mm)

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Crankshaft

A VBA program was used to compute a range of shaft diameter as shown in the appendix Table A1. A shaft length 150mm with 20mm diameter with an overhang of 10mm from the bearing #1 gives a maximum deflection of 0.22µm at the overhang and 59.51µm between the bearings. This geometry correspondingly gives a maximum shear stress of 13.5MPa and a 2350N maximum shear. These values for the material give a factor of safety of 35.8. This shaft weighs 3.62N and experiences reaction forces of 2350N and 225N respectively at the bearings #1 and #2.

The crankpin pin deflection for three pin length is shown in Figure 13 of which the 20mm pin diameter and 10mm long pin is selected for best rigidity.

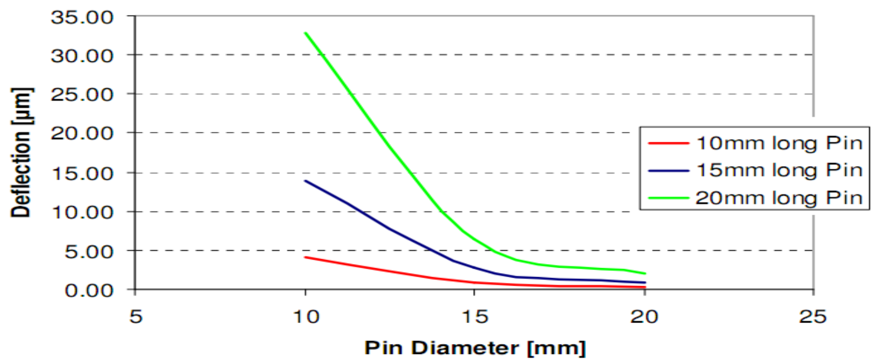

Figure 13.Crank pin deflection (for 3 sizes)

The Engine Block

The engine block is sectional shown in Figure 14. The appendix page shows a further geometric representation of the parts. More detailed design are presented in (Adeniyi, 2008).

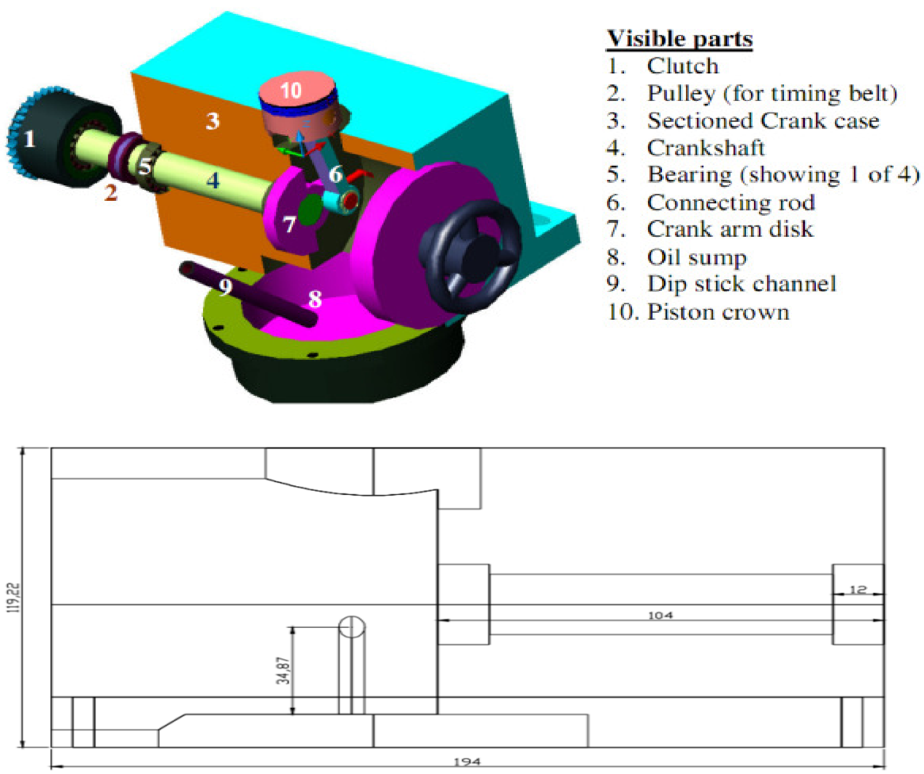

Figure 14.A section through the Engine Block

CONCLUSIONS

The Shell-Eco marathon competition is an annual competition organised to challenge young engineers to design vehicles that consume very small amount of fuel to cover extreme distances. The competition is not about how fast the vehicles can move as obtained in the Formula car race, but the fuel consumption. The drivers are allowed exhibit a driving pattern such that they can switch off the engines once the vehicle has been able to achieve acceleration big enough to allow the engine to coast for a while. During the coasting the drivers may switch off the engine to save fuel. Apart from the driving pattern, a simple and light vehicle engine but powerful enough to move it is required. This paper presented a simple design of a car engine from the basic theories. A novel concept to substitute for a need for an oil pump was discussed. To meet the requirements of a 3000 miles per gallon engine discussed in (Adeniyi & Mohammed, 2012), a 40cc engine with a bore of 40mm and 32mm stroke and overhang crankshaft length 150mm has been designed. The design did not cover the top part of the engine like the timing, camshafts and others but that could be selected from the market or garage to match.

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APPENDIX

Table A1. The Crankshaft simulation

Crankshaft Length	Max. Deflection, y1 µm	Max. Deflection, y2 µm @middle of shaft	Max. Shear Stress, N	Max. Stress in Crank -MPa	Factor of Safety	Reaction R1, N	Reaction R2, N	Weight of Crankshaft, N
Ds=10mm								
a=5mm								
L=50mm	0.44	-68.84	2591.99	54.00	8.94	2591.99	-471.11	0.30
L=100mm	0.44	-269.82	2344.12	54.00	8.94	2344.12	-222.93	0.60
L=150mm	0.44	-615.85	2267.29	54.00	8.94	2267.29	-145.81	0.90
a=10mm								
L=50mm	3.48	-116.01	2915.99	108.00	4.47	2915.99	-795.10	0.30
L=100mm	3.48	-427.12	2474.35	108.00	4.47	2474.35	-353.16	0.60
L=150mm	3.48	-957.40	2348.27	108.00	4.47	2348.27	-226.79	0.90
Ds=15mm								
a=5mm								
L=50mm	0.09	-13.59	2592.20	16.00	30.16	2592.20	-470.94	0.68
L=100mm	0.09	-53.21	2344.51	16.00	30.16	2344.51	-222.58	1.36
L=150mm	0.09	-121.19	2267.68	16.00	30.16	2267.68	-145.26	2.04
a=10mm								
L=50mm	0.69	-22.91	2916.22	32.00	15.08	2916.22	-794.98	0.68
L=100mm	0.69	-84.29	2474.76	32.00	15.08	2474.76	-352.83	1.36
L=150mm	0.69	-188.68	2348.88	32.00	15.08	2348.88	-226.26	2.04
Ds=20mm								
a=5mm								
L=50mm	0.03	-4.30	2592.49	6.75	71.50	2592.49	-470.70	1.21
L=100mm	0.03	-16.80	2345.07	6.75	71.50	2345.07	-222.08	2.41
L=150mm	0.03	-38.14	2268.70	6.75	71.50	2268.70	-144.50	3.62
a=10mm								
L=50mm	0.22	-7.25	2916.55	13.50	35.75	2916.55	-794.77	1.21
L=100mm	0.22	-26.63	2475.35	13.50	35.75	2475.35	-352.36	2.41
L=150mm	0.22	-59.51	2349.72	13.50	35.75	2349.72	-225.53	3.62
Ds=25mm								
a=5mm								
L=50mm	0.01	-1.76	2592.87	3.46	139.65	2592.87	-470.40	1.88
L=100mm	0.01	-6.86	2345.78	3.46	139.65	2345.78	-221.43	3.77
L=150mm	0.01	-15.52	2269.75	3.46	139.65	2269.75	-143.52	5.65
a=10mm								0.30
L=50mm	0.09	-2.97	2916.98	6.91	69.83	2916.98	-794.51	0.60
L=100mm	0.09	-10.89	2476.10	6.91	69.83	2476.10	-351.75	0.90
L=150mm	0.09	-24.27	2350.81	6.91	69.83	2350.81	-224.58	
Min	0.01	-957.40	2267.29	3.46	4.47	2267.29	-795.10	0.30
Max	3.48	-1.76	2916.98	108.00	139.65	2916.98	-143.52	0.60

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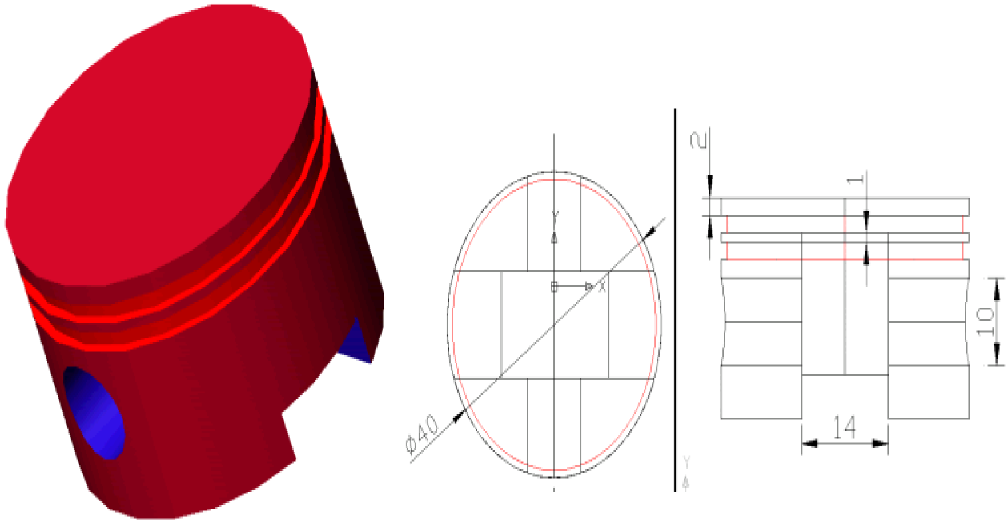

Figure A2. The Piston Geometry (Material: Aluminium)

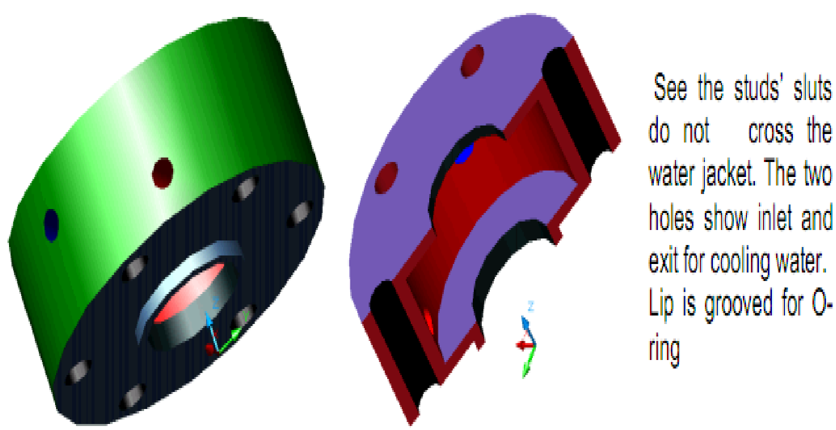

Figure A3. Top cylinder water jacket

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